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MULTI-STAGE HELICAL SCREW ROTOR

This application is a continuation-in-part of U.S. Application Serial No. 09/691,009, filed October 18, 2001, now abandoned.

Background of the Invention

The present invention relates to the vacuum pump arts. It finds particular application in a helical screw rotor vacuum pump.

Screw vacuum pumps include two pairs of helical rotors attached to shafts which are driven at high speed by an electric motor positioned below the shafts. The rotors have a plurality of teeth on their edge or arrayed on one or both of their faces and, in use, the teeth rotate within a pumping chamber and urge molecules of gas being pumped through the pumping chamber.

A gearbox is usually positioned at the driven end of each shaft. The gearbox contains the shaft ends, bearings within which the shaft rotates, any timing gears and the motor positioned about the driven shaft.

Oils and/or greases associated with lubrication of the gearbox need to be contained and isolated within the gearbox. This is to ensure cleanliness and prevent non-contamination of the gases being pumped in the pumping chamber and to avoid the possibility of transfer of such contamination back into the enclosure being evacuated.

The conventional screw vacuum pump has working rooms for compressing fluid (gas) by decreasing its volume and working rooms which have no compression action on the fluid, but has merely a fluid feeding action. Therefore, in the conventional screw vacuum pump, the pressure rises up locally (at the portion which has the compression action), and this local rise-up of the pressure causes an abnormal temperature increase at parts of the rotors and the casing of the vacuum pump. That is, the temperature at the discharge side at which the working room reduces its volume and thus compresses the gas tends to abnormally rise up. As a result, the members constituting the screw vacuum pump are un-uniformly thermally expanded due to the local temperature increase, and thus the dimensional precision of the gap between the casing and the rotors and the engaging portion's gap between the male rotor and the female rotor cannot be set to a high value.

In some prior art screw vacuum pumps, pressure adjustment devices are provided on the lower surface of the casing and in the axial direction of the rotors in order to prevent excessive rise-up of the pressure of the working rooms and thus prevent the abnormal temperature rise-up of the vacuum pump when the vacuum pump works in a state where the suck-in pressure is substantially equal to the atmospheric pressure.

Minimizing power consumption in the pump is an on-going challenge. Existing pump systems include suction sections at the ends of the rotors adjacent the closed end plates. The roots portions are provided at each of the both ends of the screw gear portions; that is, they are provided at both the suck-in side and the discharge port. A roots stage is needed adjacent the end plates. Including the suction sections at the ends of the rotor results in a less efficient compression and a smaller reduction in temperature. The roots portions of the existing pumps are difficult to machine and do not result

in an appreciably larger volume of gas being trapped and accordingly result in less efficient compression.

Accordingly, it is considered desirable to develop an improvement to the power consumption of the pump condition which would reduce power needs at high pressures and reduce rotor sizes, which would overcome the foregoing difficulties and others while providing better and more advantageous overall results.

Summary of the Invention

In accordance with a first aspect of the present invention, a vacuum pump includes a pump chamber in which an inlet and exhaust port are defined. First and second rotors are mounted parallel to each other in the pump chamber adjacent the inlet and outlet ports. A lobe is mounted to the first rotor adjacent the inlet port and a channel is defined in the second rotor adjacent the inlet port. The lobe and channel cooperate to form a suction section adjacent the inlet port.

In accordance with another aspect of the present invention, a method is provided for reducing the power consumed to move a volume of gas through a vacuum pump. A first shaft section is defined extending from a first rotor in a pump chamber adjacent an inlet port. A second shaft section is defined extending from a second rotor adjacent the inlet port. A lobe is provided on the first shaft section and a channel is defined in the second shaft section. The channel matingly engages the lobe to form a suction section between the rotors and the inlet port.

One advantage of the present invention is that it reduces power needs at high pressures, thus improving pump efficiency.

Another advantage of the present invention is that it reduces the temperature within the pump chamber due to lower power consumption.

Another advantage of the present invention is that it allows reduction in size of the rotors, thus reducing production costs.

Still another advantage of the present invention is that it reduces pump operating costs.

Yet still another advantage of the present invention is that providing the insert at the center of the screw rotors instead of at the ends of the rotors reduces machining costs.

Still other advantages and benefits of the invention will become apparent to those skilled in the art upon a reading and understanding of the following detailed description.

Brief Description of the Drawings

The invention may take form in various components and arrangements of components, and in various steps and arrangements of steps. The drawings are only for purposes of illustrating preferred embodiments and are not to be construed as limiting the invention.

FIGURE 1 shows a side elevational cross-sectional view of the existing screw vacuum pump assembly.

FIGURE 2 shows a top elevational view of the existing screw vacuum pump.

FIGURE 3 shows a perspective view of a pair of rotors with the suction sections in accordance with the preferred embodiment of the present invention.

FIGURE 4 shows a perspective view of a pair of rotors with the suction sections in accordance with a second preferred embodiment of the present invention.

FIGURE 5A shows an elevational view of a screw rotor with a widened center gap.

FIGURE 5B shows a cross-sectional view of a rotor with a widened center gap.

FIGURE 6A shows an elevational view of a screw rotor with a V-shaped male lobe in the center gap.

FIGURE 6B shows a cross-sectional view of a screw rotor with a V-shaped male lobe in the center gap.

FIGURE 6C shows an elevational view of a screw rotor with a V-shaped female portion in the center gap.

FIGURE 7A shows an elevational view of a screw rotor with a radius-shaped male lobe in the center gap.

FIGURE 7B shows a cross-sectional view of a screw rotor with a radius-shaped male lobe in the center gap.

FIGURE 7C shows an elevational view of a screw rotor with a radius-shaped female portion in the center gap.

FIGURE 8 is a graph of thread pressure vs. thread volume without internal compression.

FIGURE 9 is a graph of thread pressure vs. thread volume with internal compression at the ends of the rotors.

FIGURE 10 is a graph of thread pressure vs. thread volume with internal compression at the center gap of the rotors.

FIGURE 11 is a graph of theoretic power vs. inlet pressure.

FIGURE 12 is a perspective view of a pair of rotors with suction sections in accordance with another embodiment of the present invention.

FIGURE 13 is a top view of the rotors of FIGURE 12.

Detailed Description of the Preferred Embodiments

With reference to FIGURE 1, an existing screw vacuum pump comprises a vacuum pump **10** comprising a pump chamber **12** having a first end **13**, a second end **15**, a third end **17** and a fourth end **19**. The pump chamber **12** further comprises a central inlet port **14** located at the third end **17** of the chamber **12**, through which gas from an enclosure (not shown) connectable to the inlet can be pumped to a

pump high pressure exhaust port **16** located at the fourth end **19**.

The chamber further includes a first pair of rotors **18, 20** located within the chamber adapted for high velocity rotation horizontally within the chamber. The first pair of rotors **18, 20** are mounted on a first shaft **30** extending through the chamber **12** and into bearing mounts **32, 34** located at opposite ends of the shaft **30**. The bearing mounts **32, 34** are substantially isolated from the chamber by means of seals **42, 40**, respectively, which are mounted on the shaft **30** and located on opposite ends of the shaft **30**.

The rotors **18, 20** have teeth **44, 46**, respectively, which when mated with a second set of rotors **52, 54** (shown in FIGURE 2) create a plurality of closed chambers or cells **47** in the pump chamber **12** and urge molecules of gas to be pumped through the cells. The rotors each have low pressure inlet faces **48, 50** through which the inlet gas enters the rotor from the inlet port **14**. The teeth **44** on the rotor **18** advance in an opposite direction from the teeth **46** on rotor **20** by virtue of opposite helix direction, thus moving the gas in an opposite direction.

Referring now to FIGURE 2, the second pair of rotors **52, 54** are mounted on a second shaft **60**, which is parallel to the first shaft **30**. The second shaft **60** includes a bearing mount **62** and a seal **66** at one end of the shaft and a bearing mount **64** and a seal **68** at the opposite end of the shaft. The rotors **52, 54** have teeth **70, 72** which also advance in opposite directions from each other. The second set of rotors **52, 54** also have inlet faces **80, 82** through which gas enters the rotors from the inlet port **14**.

The seals can be of a close tolerance but non-contact design. The seals **40, 68** are located adjacent an end plate **90** which is flush with ends **91, 93** of the rotor assemblies **18** and **52**. The seals **42, 66** are located

adjacent end plate **92** which is flush with the ends **95, 97** of the rotor assemblies **20** and **54**.

Referring again to FIGURE 1, gas enters the pump through the low pressure inlet port **14**. The gas then moves in opposite directions along the helical rotors **18, 20, 52, 54** toward exhaust ports **86, 88** which are located at the first and second ends **13, 15** of the pump chamber **12** at end plates **90, 92**, respectively. End plate **90** is located at end plane **100** and end plate **92** is located at end plane **102**. The gas is essentially captured between the teeth of rotors **18, 20, 52, 54** and the fixed volume of gas is moved along the rotors **18, 20, 52, 54** toward the opposite end planes **100, 102**. Rotors **18** and **52** move the gas toward end plane **100**. Rotors **20** and **54** move the gas toward end plane **102**. As the rotors are rotated on shafts **30, 60**, the threads of the rotor threads move toward the end planes **100, 102**. The seals each include a stationary side **98, 104, 106, 108**, respectively, which are pressed into the end plates **90, 92**.

Referring again to FIGURE 2, the teeth **44** of the rotor **18** mesh with the teeth **70** of rotor **52** and push the fixed volume of gas toward the end plane **100**. The teeth **46** of rotor **20** mesh with the teeth **72** of rotor **54** and push another fixed volume of gas in an opposite direction toward the end plane **102**.

A motor **110** drives the shafts **30, 60**. Referring to FIGURE 2, the motor **110** is located beneath gearboxes **120, 122** at the motor drive end **112**. The bearing mounts **32, 34, 62, 64** surround the shafts **30, 60** and house bearings within which the shafts **30, 60** rotate. Referring to FIGURE 1, On the motor drive end **112** of the shafts, there is a pair of angular contact bearings **114, 116** which position the shafts radially and hold them in place axially in the pumping chamber. On the opposite side of the shaft is a single ball bearing **130** which also provides radial and axial support for the shafts.

As the gas enters the two exhaust ports **86, 88**, it is transported to a first exhaust cavity **126** located at exhaust port **86** and to a second exhaust cavity **128** located at exhaust port **88**. The first and second exhaust cavities lead to a third exhaust cavity **132** through which the gas flows into the high pressure exhaust port **16**.

Referring to FIGURE 3, rotors **18, 20, 52, 54** have screw thread sections **19, 21, 53, 55**, respectively, which extend in opposite directions from the center of the rotors. At the center of the rotors **18, 20, 52, 54** are center shafts **140, 150** which are positioned below the inlet port **14** within the pump chamber. The shafts **140, 150** are positioned in the center gaps of the rotors. The center gaps have been increased in width to form the shafts **140, 150**.

A preferred embodiment of the present invention comprises the shaft **140** having a raised relief male lobe **142** and a female channel **143** which is 180° opposite to the lobe **142** and is the negative profile of the lobe. Lobe **142** engages a correspondingly hollow female or channel portion **152** in the second shaft **150**. Shaft **150** also has a lobe **153** which is 180° opposite channel **152** and is the negative profile of the channel. The male lobe **142** and the corresponding female portion or channel **152** are shown to be V-shaped in FIGURE 3. The lobe **142** and channel **152** form a suction section **154**. Channel **143** and lobe **153** also form a suction section opposite section **154**.

However, in a second preferred embodiment, shafts **170** and **180** include a male lobe **172** and a female channel **182** which are round or radius-shaped as shown in FIGURE 4. This radius (R) may be increased up to and including R is equal to infinity; in which case, the leading edge of the insert would be a straight line. This straight line may b parallel to the shaft centerline. The lobe **172** and channel **182** form a suction section **184**. Similarly, shaft **170** also includes a channel **173** which is 180° opposite lobe **172** and shaft **180** includes a lobe **183**

which is 180° opposite channel **182**. There are other embodiments of the suction sections including multi-lobed suction sections which are not shown.

As seen in FIGURE 1, the existing pump screws have small center gaps **160**. As seen in FIGS. 5A and 5B, the modification to the screw rotors includes increasing the width of the center gap shaft **190**. As shown in FIGS. 6A, 6B, and 6C, a V-shaped insert is added to the center gap to forming male lobe **142** and correspondingly female channel **143** in shaft **140**. FIGURE 6C illustrates female channel **152** in shaft **150** and correspondingly male lobe **153**. FIGS. 7A and 7B show a radius-shaped lobe **172** and female channel **173** in shaft **170**. FIGURE 7C shows a corresponding radius-shaped female channel **182** and lobe **183** in shaft **180**.

FIGURE 3 illustrates the interaction of the male lobe **142** and the female channel **152**. Gas is sucked in through the inlet port **14** into the shaft sections **140, 150** and is compressed by the male lobe **142** and the female channel **152**. At the initial stage, the suction section **154** increases in volume as the rotors rotate, drawing gas into the pumping chamber. At the point where shaft **150** reaches maximum volume, a position equivalent to that shown for shaft **140** in **FIGURE 3**, the male lobe closes the suction section **154** to the inlet opening. With further rotation, the male lobe compresses the trapped suction gas into the adjacent screw section(s). The gas tightness of the suction section **154** is kept by the male lobe **142** and the female channel **152**. The increase in compression of the gas resulting from the suction sections reduces the amount of power consumed to move a volume of gas through the pump.

Under normal vacuum operation, the power consumption is predominately determined by the rotor diameter and the screw pitch at the exhaust ends of the rotor. With the increased intake volume created by the suction section, the screws are supercharged, moving a

considerably higher quantity of gas, determined by the selected volume ratio (V_r), with the same power consumption. The amount of power saved is illustrated in FIGURE 10.

FIGURE 8 is a graph illustrating power needed to move a volume of 100 cubic meters of gas per hour through the screw rotor without any internal compression. That is, the area within the curve is theoretical power consumed (3kW of power) at an inlet pressure (Pi) of 10 mbar and an exhaust pressure of 1100 mbar. The built-in volume ratio V_r is equal to 1 (one) since there is no internal compression. That is, the volume ratio is equal to the volume of gas trapped in the first screw thread at the inlet versus the volume of gas trapped in the last screw thread at the exhaust. Since there is no internal compression, the ratio is equal to 1. The cycle proceeds as follows. From state 0 to state 1, the volume of the thread is increasing with rotation of the rotor. At state 1, the first thread is closed to the inlet port. From state 1 to state 2, the closed thread advances from the inlet end to the exhaust end with the corresponding increase in pressure and without any reduction in volume. At state 2, the thread is opened to the exhaust plane. From state 2 to state 3, the transported gas is expelled from the pump. This amount of power is roughly equivalent to that which would be consumed by a roots blower or by a screw pump to move a volume of gas without internal compression (i.e., without any end plates).

Referring now to FIGURE 9, the graph illustrates that a power savings is obtained when internal compression is added to the pump at the exhaust ends of the pump cavity.

The gas begins entering the pump chamber at state 0. This continues until maximum volume is achieved at state 1. From state 1 to state 2, the gas is transported from the inlet end to the exhaust end without any reduction in volume. At state 2, the thread is not immediately exposed

to the exhaust by virtue of a close clearance end plate with a timed exhaust opening. From state 2, the thread arriving at the end plane is compressed against the end plate until the time when it is exposed to the exhaust opening at state 3. Depending on the thread pressure realized at state 2, and the selected V_r, there may be an over compression or under compression at state 3 (a slight over compression is shown). Upon exposure to the exhaust port, the thread pressure instantaneously achieves exhaust pressure (state 4). From state 4 to state 5, the gas is expelled from the pump.

The compression power needed to move a 100 cubic meter volume of gas per hour is 2.7 kW which is an approximately 10 percent savings in power from when there is no internal compression (3 kW of power). The built-in volume ratio (V_r) is 1.7. That is, the ratio of volume trapped in the first screw thread is 1.7 times the volume of gas trapped at the last screw thread at the exhaust.

In FIGURE 10, the graph illustrates the power savings due to internal compression which occurs in the preferred embodiment of the present invention. In the present invention, the internal compression occurs at the center gaps below the inlet port as the gas is pumped into the opposite screw sections. This results in an over 50 percent reduction in power consumed as compared to the power and when there is no internal compression. That is, the power consumed to move 100 cubic meters of gas per hour through the pump chamber to the exhaust is 1.3 kW as compared to 3 kW without internal compression. The built-in volume ratio V_r is 2.3. That is, the ratio of volume trapped in the suction section **154** is 2.3 times the volume trapped at the last screw thread at the exhaust.

FIGURE 11 illustrates various types of theoretical power versus inlet pressure. Isochoric pressure is shown which is pressure with constant volume pumping. Adiabatic pressure is shown which is pressure without heat exchange with the surroundings. The

isothermal curve reflects power consumed when there is no change in temperature.

A fixed V_r of 3 allows more power to be saved at low inlet pressure. That is, the higher the volume ratio, the more power is saved. Thus, at a V_r of 2.3 (corresponding to FIGURE 10) where internal pressure occurs at the center gap, additional power is saved than where internal compression occurs at the end of the rotors (V_r = 1.7, FIGURE 9). By varying the width of the center gap, the volume ratio can be altered thus changing the power consumption.

As the volume is compressed, the temperature within the pump chamber increases. When the volume is compressed at the end of the rotors, the temperature rises at the ends of the rotors. Since the volume is gradually compressed, the heat within the screw is distributed over the length of the screw. With the preferred embodiment of the present invention, since less power is needed to move the volume of gas, there is less temperature increase in the pump chamber.

With reference to FIGURES 12 and 13, a first rotor **218** includes a series of helical threads or teeth **244**. A first shaft section **240** extends from an end of the helical threads adjacent an inlet port. A second rotor **254** defines a second set of helical threads or teeth **270** which mesh with the helical threads **244** of the first rotor. As the first and second rotors rotate, the helical threads pump gases from an inlet port, along their length, to an exhaust port adjacent an opposite end thereof. The second rotor **254** has a second shaft portion **250** extending from an inlet port end thereof. The first shaft portion **240** carries a lobe **242** which is received in a complementary channel **252**. The second shaft section **250**, 180° displaced from the first lobe and channel arrangement, defines a lobe **242'** and the first shaft portion **240** defines a channel **252'**.

There are various ways the power consumption can be altered by the suction sections. The width of the center gap can be altered. Secondly, the shape of the male and female lobe connections can be varied by different geometric configurations. Third, a multi-lobed configuration could be used in lieu of a single-lobed configuration.

The invention has been described with reference to the preferred embodiments. Obviously, modifications and alterations will occur to others upon a reading and understanding the preceding detailed description. It is intended that the invention be construed as including all such modifications and alterations insofar as they come within the scope of the appended claims or the equivalents thereof.